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Comparison of the damping properties of different strip steels at different frequencies

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Abstract

The flapper valve is one of the most critical components of the reciprocating compressor. During the compressor's service lifetime, the flapper valve has to open and close billions of times without failure or maintenance. Smooth and efficient operation of the compressors not only requires an optimized design of the valve but it also places high demands on its material. Material damping of the flapper valves in the reciprocating compressors positively influences the flapper valve operation by directly reducing the magnitude of the induced stress waves in the valve reeds. It has also been reported in several studies that material damping reduces the amplitude of fatigue stresses in components or specimens subjected to fatigue load conditions.

The current study presents an investigation of the material damping properties of a range of hardened and tempered martensitic steels including the new flapper valve steel grade, Flap-X, developed by voestalpine Precision Strip AB. The tested materials included three flapper valve steel grades Flap-X, AISI 420 (SS716) and AISI 1095 (20C) along with two other thin strip materials: the S-Coat L+ tool steel and 13Cr 0.7C martensitic stainless steel grade known as UHB AEB-L. Material damping was measured using impulse excitation apparatus that measured the resonant frequencies and the loss rate of vibrations in the flexural mode in order to evaluate the material damping parameter (Q^{-1}) at room temperature. Different frequencies were investigated in this study ranging from 50 Hz to approximately 10 kHz for all the tested materials. The measured material damping data for the tested materials showed that damping decreased with increasing frequency above 50 Hz up to around 2000 Hz and then stabilized until approximately 9000Hz. Material damping for the Flap-X grade was found to be higher than the SS 716 grade at all the measured frequencies and higher than all the tested grades at ~9000 Hz. The material damping of the 20 C grade was found to be the highest at 50 Hz while the UHB AEB-L grade displays the highest damping at 250 Hz and 2000 Hz. In addition, an attempt has been successfully made to fit the Rayleigh damping model to the obtained damping data. These results have significant implications for the impact fatigue stresses in the valves, their impact fatigue life and the noise levels they contribute to in the reciprocating compressors.

1. Introduction

The flapper valve is one of the most critical components of the reciprocating compressor. During the compressor's service lifetime, the flapper valve has to open and close billions of times without failure or maintenance. Smooth and efficient operation of the compressors not only requires an optimised design of the valve but it also places high demands on its material. Through extensive experience and extensive testing over the years, voestalpine Precision Strip AB has developed the steel grades that can meet the stringent demands of the flapper valve. The developed steel grades possess high tensile strength, bending and impact fatigue strength, controlled non-metallic inclusion content, fine surface finish and good flatness; see Löf et al. (2016). These characteristics allow efficient and reliable operation of the flapper valves and ensure long service life of the compressors without need for maintenance or replacement of the valve.

Damping is another crucial material property that can have a significant effect on the service fatigue life and might also facilitate less noisy operation of the compressor. High material damping reduces the amplitude of vibrations that are induced in the valve reeds on impact with the valve plate. It has also been reported in several studies that material damping reduces the amplitude of fatigue stresses in components or specimens subjected to fatigue and impact fatigue load conditions, see Kazymyrovych et al. (2010) and Chai et al. (2004). Therefore, a detailed investigation of the damping properties of the flapper valve steels would aid the design of more robust flapper valves and reduce the noise generated during the operation of a compressor.

This study aims to investigate the damping characteristics of different flapper valve steel grades and compare their performance with respect to damping with each other and to the other steel grades. For this purpose, an impulse excitation apparatus was used to measure the loss rate, resonant frequencies and material damping at different frequencies of interest.

2. Materials and specimens

In this specific study, a range of strip steel grades was investigated for their damping properties. Flap-X and SS 716 steel grades are upgrades of the standard martensitic stainless steel AISI 420. The “SS 716” grade is a well-established valve steel which has been used for many years while “Flap-X” is the latest upgraded version developed and produced by voestalpine Precision Strip AB. The “20C” steel grade is a carbon valve steel grade. The S-Coat L+ is a tool steel grade while UHB AEB-L is a razor blade steel grade. The chemical composition of the tested grades is shown in the table below:

Table 1: Nominal chemical composition of the tested steel grades (wt. %)

Steel grade	C	Si	Mn	Cr	Mo	N	V	P	S
SS 716	0.38	0.45	0.55	13.5	1.00			≤0.025	≤0.015
Flap-X	0.38	0.45	0.56	13.5	1.00	+	+	≤0.025	≤0.015
20 C	1.0	0.3	0.40	0.15	-	-	-	≤0.025	≤0.015
S-Coat L+	0.55	1.0	0.75	2.6	2.25	-	-	-	-
UHB AEB-L	0.68	0.4		12.8	0.6	-	-	≤0.025	≤0.015

The tested valve steel grades are high strength hardened and tempered strip grades that exhibit martensitic microstructures. The materials are hardened and tempered at suitable temperatures and times in order to obtain good blend of mechanical strength and toughness. The mechanical and physical properties of all the tested steel grades are shown in Table 2. Flap-X possesses higher ultimate tensile strength than the rest of the tested grades. S-Coat L+ and UHB AEB-L can be delivered in a range of different tensile strengths.

Table 2: Mechanical and physical properties of the tested steel grades

Steel grade	Thickness (mm)	Elastic modulus (GPa)	Density (kg/m^3)	Tensile strength (MPa)
SS 716	0.381	210 – 220	7700	1810±80
Flap-X	0.381	210 - 220	7700	2100±60
20 C	0.381	210	7850	1860±50
S-Coat L+	0.381	210	7800	-
UHB AEB-L	0.305	220	7800	-

The specimens tested in this study were cut longitudinally along the rolling direction of the materials. The specimen design was based on equation (1) for calculating the flexural resonance frequency f_r:

$$f_r = \sqrt{\frac{Eb}{0.9465 * m * \left(\left(\frac{t}{L}\right)^2 * 0.6858 + 1\right) * \left(\frac{L}{t}\right)^3}} \qquad (1)$$

where E and m represent the elastic modulus and mass of the specimens while *b*, L and t represent the width, length and thickness of the specimens, respectively. Four different rectangular strip specimen designs were selected that possessed four different resonance frequencies, as shown in Table 3.

Table 3: Four different specimen designs of Flap-X steel grade selected for four different resonance frequencies

Frequency (Hz)	Length (mm)	Breadth (mm)	Thickness (mm)
51	200.0±0.05	6.0±0.05	0.381
253	90.0±0.05	6.0±0.05	0.381
2278	30.0±0.05	6.0±0.05	0.381
9097	15.0±0.05	6.0±0.05	0.381

The values for resonance frequencies reported in Table 3 are calculated from equation (1) by using E=213.5 GPa and ϱ (density) = 7700 kg/m3. Care should be taken to maintain consistency of units when using equation (1).

For each design, three separate specimens were taken from different parts of the strip and 15 repetitions of measurements were performed on each specimen. The results presented in the later sections give the average value of 45 repetitions for each steel grade at each resonance frequency.

3. Method

In this study, an impulse excitation apparatus was used to determine the resonant frequencies and damping values. The principle of this method and more detailed description has been explained by Roeben et al. (1997). In this apparatus, specimens of specific design as determined by equation (1) were placed freely on two supports (nylon strings) such that the nodes of vibration in the specimens lied directly over the supports. The specimens were manually impacted near the middle (antinode of vibration) with a steel ball attached to a flexible polymeric rod. The specimen vibration was recorded by a microphone, see Figure 1a, and the signal was stored on a computer.

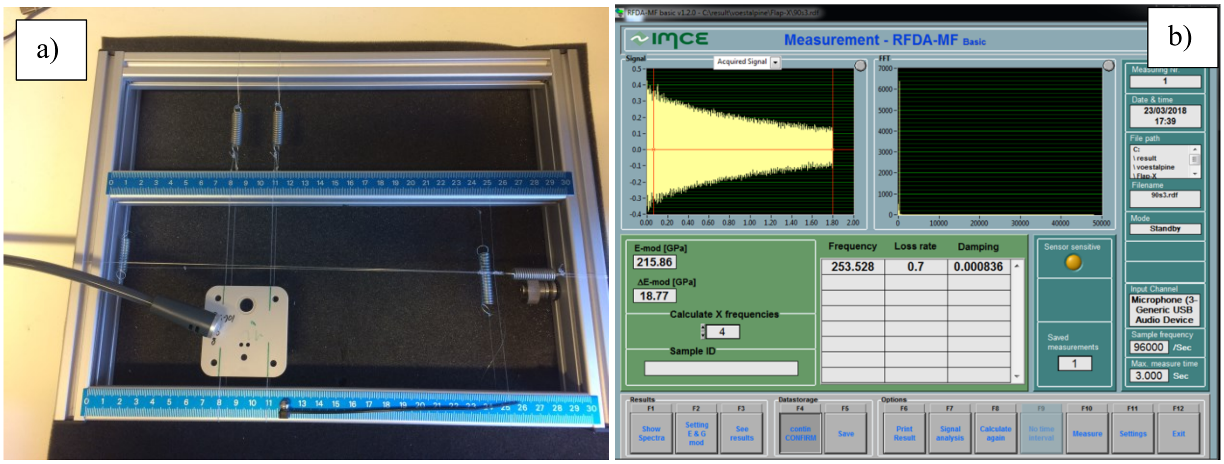

Figure 1: a) Impulse excitation apparatus with a rectangular sample placed on two parallel threads where nodes of flexural natural frequency lie, b) RFDA.MF software measures the natural frequency, elastic modulus and damping based on decay of the amplitude of vibrations.

A program written in LabVIEW^{TM} determines the resonant frequency f_r by Fourier analysis. The program assigns a vibration of the form indicated by equation (2) to each resonant frequency f_r.

$$x(t) = Ae^{-kt}\sin(2\pi f_r t + \varphi) \tag{2}$$

Where A, k, t, φ represent the initial amplitude of vibration, loss rate, instantaneous time and phase angle, respectively. For each value of k the corresponding damping is calculated using equation (3):

$$Q^{-1} = \frac{k}{\pi f_r} \tag{3}$$

where Q^{-1} gives a value for internal material friction or damping.

1.1 Rayleigh damping model

The Rayleigh damping model, see Rayleigh (1877), is widely used to compute damping for structural vibration problems. It assumes that the viscous damping is proportional to a linear combination of mass and stiffness. For a multi-degree of freedom system, the damping matrix "C" is given by the expression:

$$C = \alpha M + \beta K \tag{4}$$

where M = mass matrix

K = stiffness matrix

α = mass proportional constant

β = stiffness proportional constant

Another feature of Rayleigh damping is that it varies with the response frequency according to the relationship in equation (5):

$$\zeta = \frac{\alpha}{2\omega} + \frac{\beta\omega}{2} \quad (5)$$

where ζ = damping ratio

$\omega = 2\pi f$ = angular frequency

In equation (4), the mass proportional damping contributes inversely with response frequency while the stiffness proportional damping contributes linearly with the response frequency.

4. Results

Damping showed a decreasing trend with increasing resonance frequency up to approximately 2000 Hz for the tested steel grades, as shown in Figure 2. However, after approximately 2000 Hz damping showed a slight increase for some of the tested grades. In case of Flap-X, this increase is much more noticeable. It should be noted again that the data points plotted in Figure 2 are the average values (3 samples, 15 measurements on each sample) at each resonance frequency. There is some scatter in the measured damping values at each of the tested frequencies for all the steel grades.

Flap-X showed higher damping values compared to the standard SS 716 valve steel grade at all of the tested frequencies except the 50 Hz. At ~2000 Hz and ~9000 Hz the measured average damping of the Flap-X grade was higher than the other valve steel grades SS 716 and 20 C. The measured resonance frequencies for the different tested steel grades are slightly different depending on the difference in their elastic modulus, density and probably also due to slight differences in the dimensions of the prepared samples. Similar to the damping values, the frequency values plotted in Figure 2 are also the average values.

The carbon steel grade 20 C used for valve applications showed the highest damping measured at 50 Hz and displayed higher damping than other valve steels at ~250 Hz. The razor blade steel grade UHB AEB-L displayed the highest damping at ~250 Hz and ~2000 Hz frequencies.

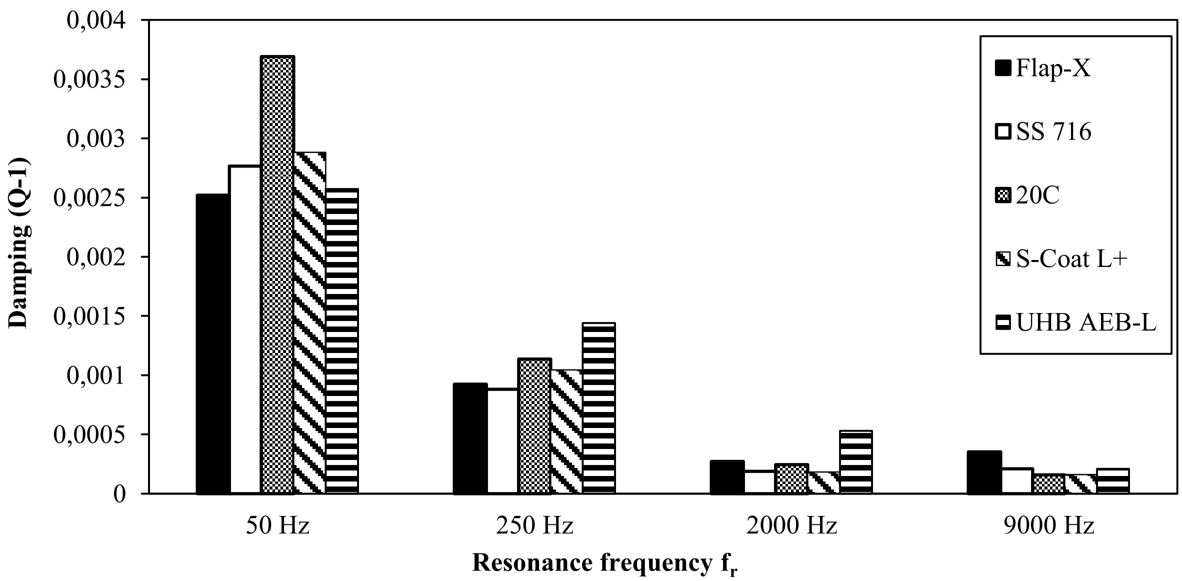

Figure 2: Damping measured at different resonant frequencies for different strip steel grades.

The material damping parameter presented in this paper (Q^{-1}) is a commonly used measure of damping. This parameter also has an added advantage that it can be readily converted to other damping parameters such as critical damping ratio (i.e. using $\zeta = Q^{-1}/2$) and the logarithmic decrement (i.e. using $\tan\delta = Q^{-1}$).

The Rayleigh model was fitted to the damping measurements conducted in this study for Flap-X as shown in Figure 3. The values of $\alpha = 2.646$ and $\beta = 5.49\text{e-}9$ were computed by interpolating the damping ratio values measured at 9221 Hz and 2311 Hz. It can be seen that the Rayleigh damping model fits very well with the damping measurements especially at higher frequencies. The Rayleigh damping model captures the variation of the damping ratio with respect to the resonance frequency very well. There is some deviation from the fitted model at the lower frequencies and that is probably because the parameters α and β were computed from the damping ratios measured at the higher frequencies.

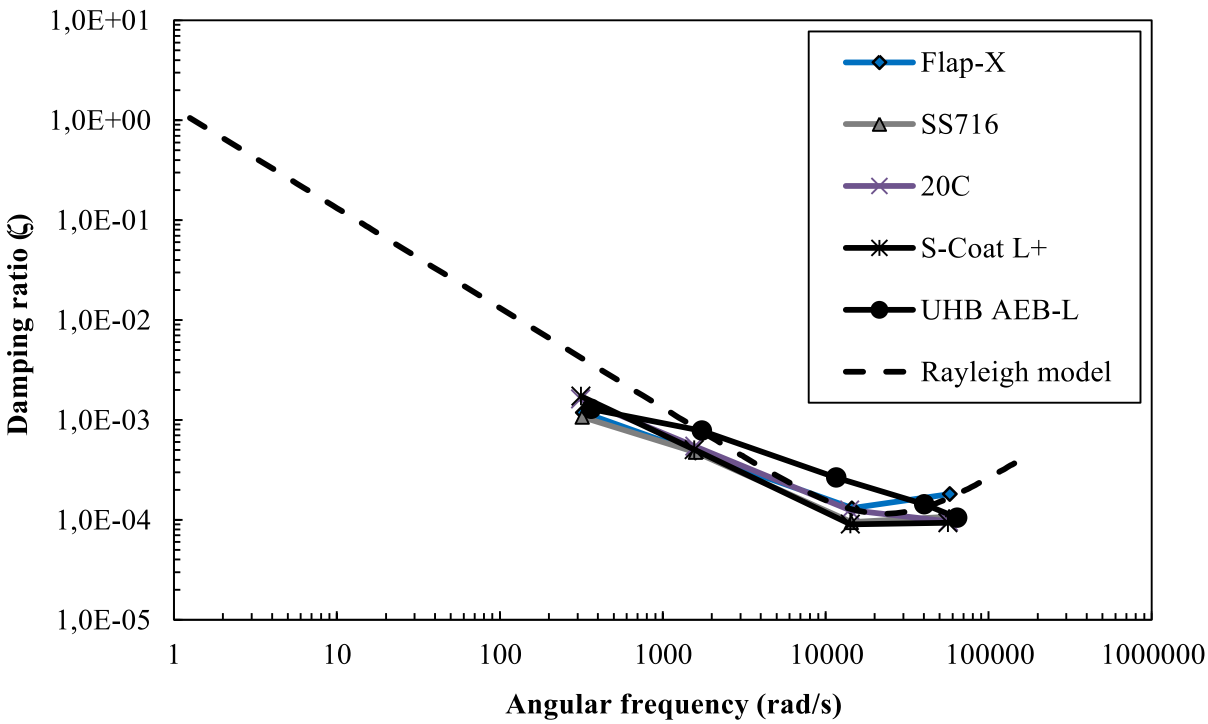

Figure 3: Fitting the Rayleigh damping model (using $\alpha = 2.646$, $\beta = 5.49\text{e-}9$) to the damping measurements for the Flap-X grade.

5. Discussion

In flapper valves, the higher material damping might help to reduce the noise level of operation by reducing the amplitude of vibrations generated due to the impact of valve reed against the valve plate. In addition, when the reed valve impacts the valve seat impact fatigue stresses are generated in the form of waves. These stresses are responsible for the failure of reed valves in most cases. The decay rate of these stresses depends on the damping of the reed valve material as explained by Chai et al. (2004). Higher damping capacity of the reed valve material is a key factor to the faster decay of these stress waves. Svenzon et al. (1976) displayed the correlation between higher damping capacity of the reed valve material and their higher impact fatigue strength.

This study shows a comparison of the damping capacity of the valve steel grades, especially the recently developed Flap-X grade, and some other steel grades used for other applications. The results presented in Figure 2 suggest that material damping for the tested flapper valve steels was the highest at the lowest resonance frequency that was investigated i.e. at 50 Hz. With increase in resonance frequency, the material damping showed a decrease. This result shows an important correlation between material damping and frequency. It also suggests that higher frequencies may present greater problems for the flapper valves in terms of their impact fatigue strength, impact fatigue life and noise level.

The higher damping capacity of the Flap-X grade compared to the SS 716 grade at the investigated frequencies higher than 50 Hz suggests that the flapper valves manufactured from Flap-X are likely to possess higher impact fatigue strength and longer impact fatigue life. Moreover, greater damping of induced vibrations and decay of impact fatigue stress waves could also mean higher impact fatigue strength and longer service lifetime of the Flap-X valve reeds. This correlates well with data that has already been reported in previous studies that Flap-X possesses significantly higher fatigue strength compared to SS 716 grade, see Löf et al. (2016), and other leading valve steels. On the other hand, the significantly higher damping of 20 C valve steel grade at ~50 Hz and ~250 Hz compared to the other valve steel grades indicates their suitability for the valve manufacturing applications where corrosion is not an operational limitation.

There was some deviation from the average measured values of damping reported in Figures 2, 3 at all the tested frequencies for all the steel grades. This deviation in the measured values makes the differences between the steel grades at a particular frequency less obvious. The deviation in the measured values might be due to the vibration of the nylon threads on which the samples are placed in the impulse excitation apparatus. The positioning of the microphone over the vibrating specimen also seems to play an important role in scattering of the results.

The Rayleigh model has proved to be a good tool that allows us to predict the damping ratio at the frequencies between the data points (interpolation) and outside the measured frequency range (extrapolation), see Figure 3. For instance, there are certain practical limits on the size of specimen on which the damping measurements can be made using the impulse excitation apparatus. Testing at too high frequencies (i.e. >9000 Hz) was difficult due to the length of the strip specimen that is required being too small. On the contrary, testing at too low frequencies (i.e. <50 Hz) induces curvature in the thin strip specimens (0.381 mm thickness) due to the cutting stresses. Therefore, the Rayleigh model can be relied upon to predict the damping ratio values at frequencies that are higher or lower than those that can be practically tested on such strip materials. Figure 3 shows that the Rayleigh damping model indicates that one can expect the damping ratio to increase further beyond 9000 Hz frequency. Frequencies beyond 9000 Hz were the range where Flap-X had the highest damping of all measured valve materials.

6. Conclusion

Following conclusions could be drawn from this study:

1. High damping capacity of Flap-X at the tested frequencies is a beneficial characteristic for reed valves in compressors.
2. Material damping shows a decreasing trend with increasing frequency up to approximately 2000 Hz for the tested grades. After ~2000 Hz material damping showed slight increase which was especially noticeable for Flap-X.

3. The Rayleigh damping model when fitted to the damping results predicted the variation in damping with frequency very well. Therefore, the Rayleigh damping model can be a good tool to compute the material damping values higher and lower frequencies that pose difficulties of measurements due to the specimen size.

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